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ScienceDirect
Energy Procedia 49 (2014) 637 – 646
Energy Procedia

SolarPACES 2013

Study on solid particles as a thermal medium

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Abstract

This experimental study is conducted as a part of a Department of Energy funded SunShot project titled "High Temperature Falling Particle Receiver". In this concept, solid particles are heated by concentrated sunlight to very high temperatures to the point that they can become a suitable heat source for various thermal power and thermochemical cycles. Furthermore, one of the great advantages of this concept is the ability to store thermal energy in the solid particles at relatively low cost. However, an important feature of any Particle Heat Receiver (PHR) system is the particle to fluid heat exchanger (PFHXer), which is the interface between the solar energy system and the thermal power or chemical system. In order to create this system material data is needed for the design and optimization of this PFHXer.

This study focuses on the heat transfer properties of particulates to solid surfaces. The particulates will be evaluated for three grain sizes of sand and two grain sizes of proppants. These two materials will be tested at one, five and ten millimetres per second in order to see how the various flow rates, which will be required for different loads, will affect the heat transfer coefficient. Finally the heat transfer coefficient will also be evaluated for both finned and non-finned heat exchangers to see the effect that changes in the surface geometry and surface area have on the heat transfer coefficient. The heat transfer coefficient will help determine the appropriate material that will be used in the PHR system.

An experimental procedure is under development by Georgia Institute of Technology to examine the heat transfer coefficient. To accurately characterize the size distribution of the materials, the material will be placed through a sifting unit. Afterwards the material will be placed into the testing apparatus. The main components of the testing apparatus consist of the Old's Elevator which is used to move the sand and the PFHXer. The PFHXer is made up of the constant head plenum and the heating box. The constant head plenum allows for a constant particulate head and uniform flow. The heating unit is made up of eight electric cartridge heaters. These heaters act as the non-finned tubed heaters and through the use of a metal casing can also be changed into finned tubed heaters. By running the system to steady and state and measuring the surface, inlet and outlet temperatures the heat transfer coefficient can be calculated.

Selection and peer review by the scientific conference committee of SolarPACES 2013 under responsibility of PSE AG.
Final manuscript published as received without editorial corrections.

Keywords: Heat Transfer Coefficient; Solid Particulates; Heat Exchanger; Thermal Medium; Heat Transfer; Proppants; Sand

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Final manuscript published as received without editorial corrections.
doi:10.1016/j.egypro.2014.03.069

1. Introduction

This experimental study is a smaller part of the of the "High Temperature Falling Particle Receiver" project. The project is focused on the concept of using solid particulates as a thermal medium for a concentrating solar power tower. Typically molten salts are used as a thermal medium in order to absorb heat from a focal point where solar energy is focused and then transfer the heat to the rest of the system. Unfortunately, molten salts suffer from several disadvantages associated with the corrosive the effects on the pipes, ensuring that the salts remain in a molten state and a max operating temperature of approximately 550° C [1]. On the other hand, the solid particulates that are being considered can be heated above well to theoretically 900 - 1000° C. In addition to the simple thermal advantages of reaching higher temperatures, this approach avoids problems dealing with the solidification of the molten salts within a storage unit.

To use a solid particulate as a thermal medium, both a particle heating receiver and a particle to fluid heat exchanger (PFHXer) need to be designed. The PFHXer acts as the interface between the solar energy collection system and the thermal power or chemical system. This paper focuses on the material selection for the thermal medium. Alongside material selection, this study looks into the benefits of using fins for the heat exchanger. Table 1 shows the materials that are tested along with their Sauter Mean Diameter which is calculated using the mass size averages of various mesh sizes.

Table 1: This tables shows the sauter mean diameter of the materials tested

	Fracking Sand	Atlanta Industrial Sand	Riyadh White Sand	Small Proppants	Large Proppants
Sauter Mean Diameter (mm)	0.229	0.301	0.343	0.268	0.758

The two main types of materials being considered are sand and proppants. The sands being tested mainly consists of silica with sizes varying from 1.34mm to 0.212mm. Proppants are small spherical beads made up of corundum and mullite [2]. They are most commonly used for the purposes of fracking. The sizes of these proppants vary from 1.00mm to 0.15mm.

To evaluate these materials, the heat transfer coefficient will be calculated for both a finned tube and bare tube heat exchanger configuration. The two different configurations will provide the data necessary to calculate the fin effectiveness which will show how advantageous the fins are for this project. In addition, the fin efficiency is calculated in order to easily predict the results of similar cases.

Nomenclature

$Area_{bare}$	Area of a bare tube
$Area_{base}$	Area of the base of the finned tube
$Area_{fin}$	Area of one fin
$Area_{ratio}$	Area ratio between a finned tube and a bare tube
$Area_{surface,box}$	Surface are of the heat exchanger module
$Area_{total,finned}$	Total area for one finned tube
c_p	Specific heat of the particulate
ϵ_{fin}	Fin effectiveness
η_{fin}	Fin efficiency
$h_{apparent}$	The h required for a bare tube bundle to transfer the same heat rate as a finned tube

$h_{average}$	Average heat transfer coefficient for the bare configuration
$h_{effective}$	Effective heat transfer coefficient for the finned configuration
I_0, I_1	Zero and first order Bessel function of the first kind
K_0, K_1	Zero and first order Bessel functions of the second kind
k	Thermal conductivity
Length	Length of the bare and finned tube
LMTD	Logarithmic mean temperature difference
$\dot{m}$	Mass flow rate of the particulate
N_{fins}	Number of fins
q_{heater}	Power input for one heater
q_{loss}	Total calculated heat loss from the heat exchanger module
r_1	Radius to the base of the fin
r_2	Radius of the fin
r_{2c}	Corrected radius
t	Thickness
$T_{avg,par}$	Average particulate temperature
$T_{base,bot}$	Average surface temperature of the bottom tube
$T_{base,top}$	Average surface temperature of the top tube
$T_{par,inlet}$	Particulate inlet temperature
$T_{par,outlet}$	Particulate outlet temperature
U	Overall heat transfer coefficient

2. Experimental setup

To perform this experiment two PFHXer modules are created to accommodate the finned and bare tubes. The heat exchanger module is a box with inner dimensions of 0.114m by 0.114m by 0.114m. In each box there are eight are cartridge heaters which will run through the box. These heaters are parallel to one another and are then placed in series to a watt meter and a VARIAC. The VARIAC is used to adjust the wattage in order to keep a fairly constant power level across the varying tests for the different speeds and materials which are used. The heaters are placed in rows of three, two and three as shown in Figure 1. In one case the heaters are installed as bare tubes, in the other case the tubes have fins, shown in Figure 2, securely fit over them. The bare tube box is made out of polycarbonate and the finned tube box is made out of Ultem. Ultem is a thermoplastic that is used due to its higher operating temperatures [3]. The higher operating temperature is necessary due to the fin's proximity to the walls of the heat exchanger module.


Figure 1: Typical Heat Exchanger Configuration


Figure 2: Finned Tube

The top middle and bottom middle heating units each have three thermocouples attached to them. As seen in Figure 3, each of these heaters has one thermocouple placed on the top, one on the side and one on the bottom. The averages of these thermocouples, with the side thermocouple weighted twice, are used as the surface temperatures. These thermocouples are attached to an Agilent Data Acquisition Unit in order to record the data. In the bare tube case the temperature probes are taped onto the tubes, for the finned tubes the thermocouples are soldered into place.


Figure 3: Thermocouple Placement


Figure 4: Grates that Control the Flow Rate

In order to run the particulates through the module and obtain useful data the system needs to be able to run continuously until a steady state condition has been reached. To facilitate this, the module is attached to an OLDS Elevator which acts as a lift. Unfortunately the OLDS Elevator does not run at a perfectly consistent speed. As such a wooden chute with an overflow discharge tube is placed above the module to provide a constant head plenum and to help ensure a uniform flow rate. Within the chute, prior to the inlet of the heat exchanger module, a temperature probe is placed to measure the inlet particulate temperature. In addition to the chute, a grate system is placed at the outlet of the heat exchanger module. The grate system is made up of three grates. The first two grates are stationary and form a chequered mesh as shown in Figure 4. The third layer of the grate slides horizontally in order to adjust the width of the opening. The chequered pattern of the first two grates helps prevent bridging which can cause significant problems during data collection at the slower speeds. Unfortunately, the use of grates as the method to control and provide a uniform flow causes problems when attempting to read the outlet temperature. As such an energy balance is used to calculate this value. A chute has been added after the grates to combine the flows and allow for the use of a 'bucket' test in order to measure the mass flow rate. The full setup of this experiment can be seen in Figure 5.

Figure 5: Full Setup of the System

3. Data Analysis Methods

1.1 Bare Tubed Heat Exchanger

$$\mathrm{Area}_{\mathrm{bare}} = 2\pi r_1 L \tag{1}$$

First the geometry of the heaters is taken into account. In the bare tube case the equation is a simple calculation for the surface area of the tube as shown in equation 1.

In order to calculate the average heat transfer coefficient across a bundle of tubes, the logarithmic mean temperature difference (LMTD) between the tube surface temperature and the sand bulk temperature is calculated using equation 2.

$$\mathrm{LMTD} = \frac{(\mathrm{T}_{\mathrm{base,top}} - T_{par,inet}) + (\mathrm{T}_{\mathrm{base,bot}} - T_{par,outlet})}{\ln\left(\dfrac{\mathrm{T}_{\mathrm{base,top}} - T_{par,inet}}{\mathrm{T}_{\mathrm{base,bot}} - T_{par,outlet}}\right)} \tag{2}$$

With the current setup the experiment cannot accurately measure the outlet particulate temperature. To compensate for this a control volume energy analysis is performed. To find the thermal conductivity, equations 3 to 5 are used with stagnant air being assumed and experimental values from the bare tubed case as the heat transfer coefficient of sand.

$$T_{avg,par} = \frac{T_{avg,inlet} + T_{par,outlet}}{2} \tag{3}$$

$$U = \frac{1}{R_{wall} + R_{Sand} + R_{air}} \tag{4}$$

$$UA_{surface,box} = \frac{q_{loss}}{T_{avg,par} - T_{amb}} \tag{5}$$

The UA value for the box is 0.2425 W/K which is used to implicitly find the heat loss and the outlet temperature.

$$q_{\text{loss}} = (UArea_{surface,box})(T_{avg,par} - T_{amb}) \tag{6}$$

$$T_{par,outlet} = T_{par,inlet} + \frac{q_{heater} - q_{loss}}{\dot{m}c_p} \tag{7}$$

The average heat transfer coefficient is calculated in equation 8, which represents the average heat transfer coefficient over the entire finned surface of the tube bundle. $h_{average}$ is based on LMTD between the surface temperature at the base of the fins and the sand bulk temperature.

$$h_{average} = \frac{q_{tube}}{Area_{bare,LMTD}} \tag{8}$$

1.2 Finned Tube Heat Exchanger

The geometry of the finned heaters is calculated using equations 9 through 13. The area ratio equation is used to help show the effect of the additional surface area.

$$Area_{base} = 2\pi r_1 (L - tN_{fins}) \tag{9}$$

$$r_{2c} = r_2 + \frac{t}{2} \tag{10}$$

$$Area_{fin} = 2\pi(r_{2c}^2 - r_1^2) \tag{11}$$

$$Area_{total,finned} = Area_{base} + Area_{fin} N_{fins} \tag{12}$$

Similar to the bare tube case equation 2 is used to calculate the LMTD for the finned tube case. To calculate the outlet temperature equations 3 through 7 are used with a UA value of 0.2934 W/K.

For the finned tube heat exchanger the following heat transfer formula is used to calculate the effective heat transfer coefficient and the fin efficiency.

$$q_{\text{tube}} = h_{effective} Area_{base} LMTD + h_{effective} Area_{fin} LMTD \eta_{fin} N_{fins} \tag{13}$$

The $h_{effective}$ term represents the effective heat transfer coefficient over the entire finned surface of the tube bundle. It is based on LMTD between the surface temperature at the base of the fins and the sand bulk temperature.

$$c_{\text{fin}} = \frac{2r_1}{m(r_2^2 - r_1^2)} \frac{K_1(mr_1)I_1(mr_{2c}) - I_1(mr_1)K_1(mr_{2c})}{I_0(mr_1)K_1(mr_1) - K_0(mr_1)I_1(mr_{2c})} \tag{14}$$

$$\text{m} = \sqrt{\frac{2\text{h}_{\text{effective}}}{kt}} \tag{15}$$

η_{fin} represents the ratio between heat transfer rate from the fin and that from an identical fin with an infinite thermal conductivity. The fin efficiency was found using the above analytical solution [4]. In equation 14 I_i and K_i respectively represent the i-th order Bessel Function of the first and second kind.

$$h_{apparent} = \frac{q_{tube}}{Area_{bare,LMTD}} \tag{16}$$

The previous equation uses the same q from the finned heat exchanger experiment in order to calculate the

$h_{apparent}$ which represents the heat transfer coefficient that would be required for a bare tube bundle to transfer the same heat rate for the same LMTD value as the finned tube.

$$\in_{\mathrm{fin}} = \frac{h_{effective}}{h_{average}} \tag{17}$$

Equation 17 shows the fin effectiveness which is the enhancement ratio between the heat transfer rate with the fin and the heat transfer rate without the fin for the same surface temperature. A minimum value of two is usually required for the fin to be considered effective.

4. Results

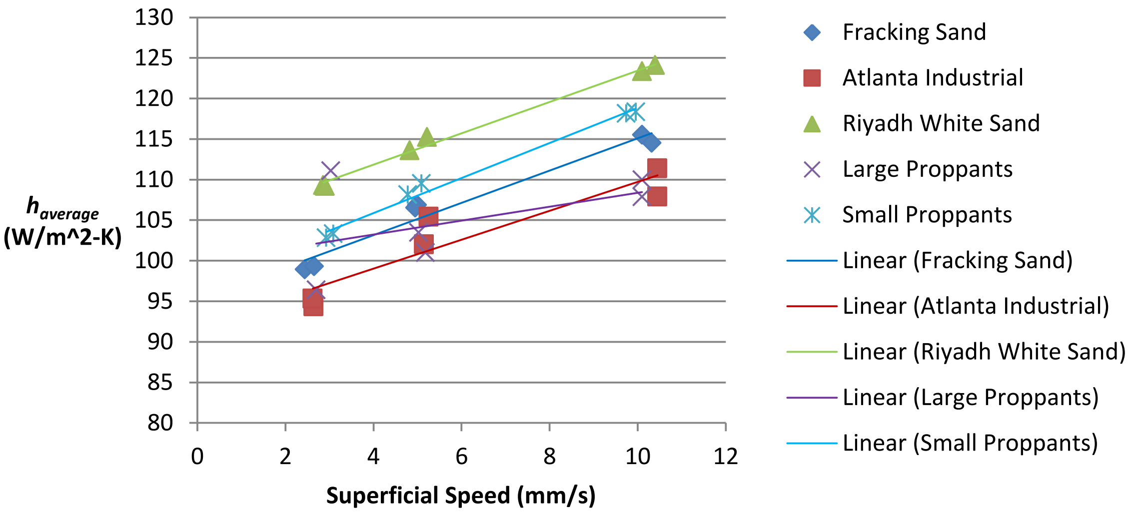

Figure 6: Bare Tubed Heat Exchanger - Average Heat Transfer Coefficient

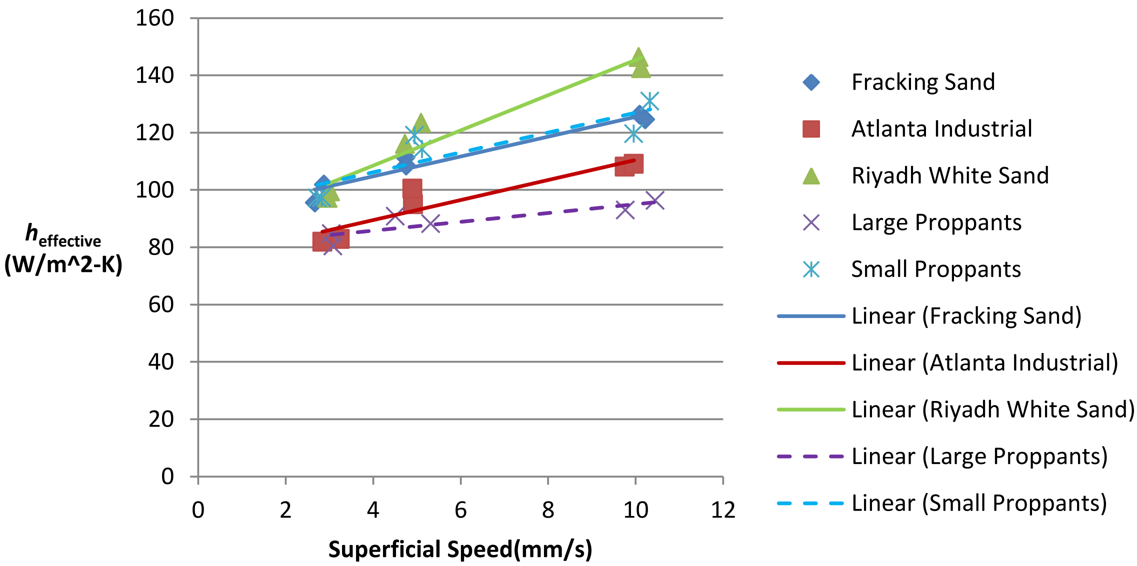

Figure 7: Finned Heat Exchanger - Effective Heat Transfer Coefficient

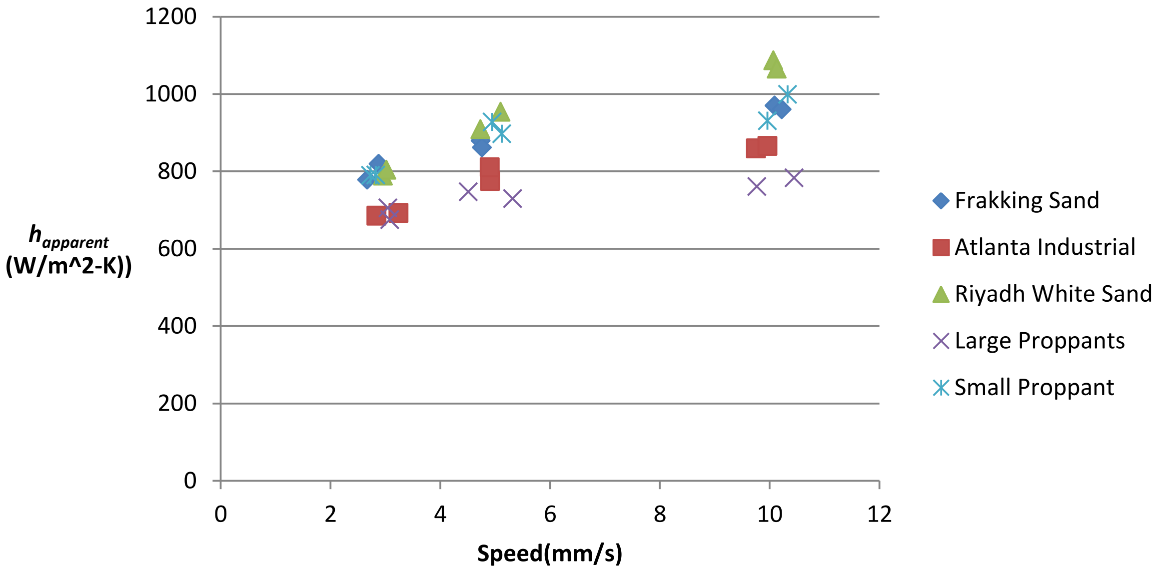

Figure 8: Finned Heat Exchanger - Apparent Heat Transfer Coefficient

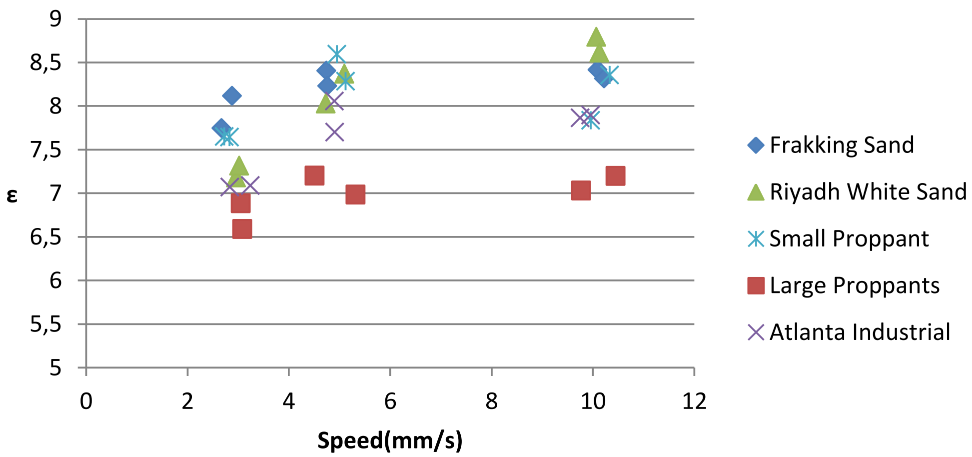

Figure 9: Finned Heat Exchanger - Fin Effectiveness

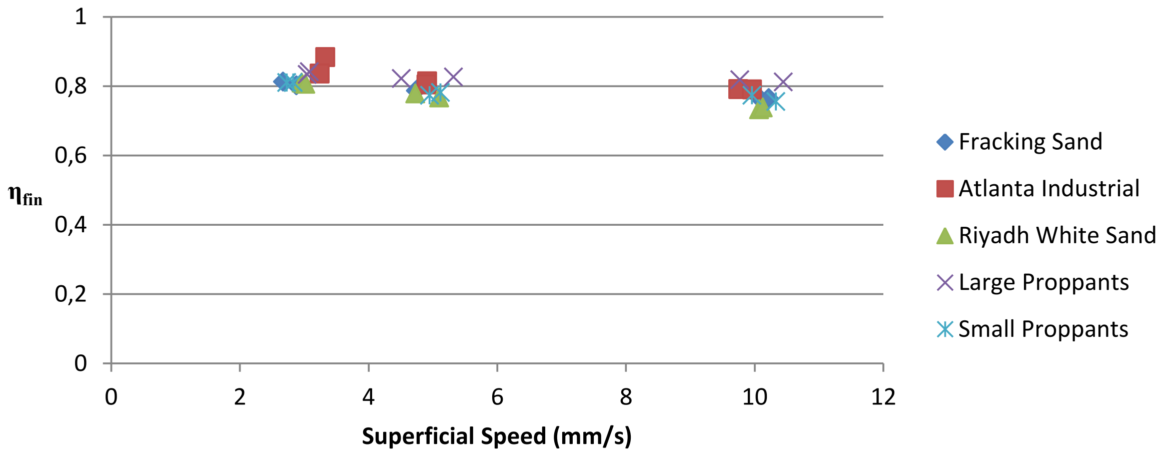

Figure 10: Finned Heat Exchanger - Fin Efficiency

5. Analysis

An important note for the data taken is that it corresponds with the superficial speed. That is, the speed is shown as if the cross sectional area is 0.1143m by 0.1143m. However, as the sand flows through the module the heaters cause a decrease in the cross sectional area and results in an increase in flow velocity near the tubes. The finned tubes have a larger profile than the bare tubes, so it is important to realize that for the same superficial speed the actual speed near the tubes is higher in the finned case than in the bare tube case.

As seen from Figures 6 and 7 the most effective material in heat transfer ability is the Riyadh White Sand. Though the Riyadh White Sand has the highest heat transfer coefficient it is also one of the more lightly colored sands which affects the absorptivity of the receiver.

A fin effectiveness value above two usually means that fins will help the heat transfer and should be used. Not only would these fins increase the heat transfer rate but relative to the price of the heat exchanger pipe, the fins are relatively inexpensive.

6. Conclusion

This paper shows the heat transfer properties of a variety of materials through both a finned and bare tube heat exchanger. The following table presents a summary of the heat transfer coefficient values found from this experiment.

Table 2: This table presents a summary of the $h_{average}$ for the bare tube heat units of W/m^2-K.

	Bare Tube Heat Exchanger					
	~3mm/s		~5mm/s		~10mm/s	
	~0.6kW	~0.9kW	~0.6kW	~0.9kW	~0.6kW	~0.9kW
Riyadh White Sand	109	109	114	115	123	124
Atlanta Industrial Sand	94	95	102	105	107	111
Fracking Sand	99	99	107	107	115	115
Large Proppants	96	111	101	103	108	110
Small Proppants	103	103	108	110	118	118

Table 3: This table presents a summary of the $h_{effective}$ for the finned heat exchanger in units of W/m^2-K.

	Finned Heat Exchanger					
	~3mm/s		~5mm/s		~10mm/s	
	~0.6kW	~0.9kW	~0.6kW	~0.9kW	~0.6kW	~0.9kW
Riyadh White Sand	97	100	116	123	146	142
Atlanta Industrial Sand	82	83	95	100	108	109
Fracking Sand	101	96	111	108	126	125
Large Proppants	80	84	88	90	93	96
Small Proppants	97	97	114	120	119	131

The Riyadh White Sand has been shown to consistently better heat transfer properties than the other materials. Additionally, the data also shows that the heat transfer coefficient directly increases with the superficial speed through the module. Unfortunately, a correlation between mean particle size and the heat transfer coefficient has been inconclusive due to both inconsistent data and the fact that each of the materials is made up of a different chemical composition. The fins are an effective addition to the tubes as shown by the fin effectiveness values that are above two.

In order to study the correlation between particle sizes and heat transfer there are plans to run this experiment using mono-disperse glass beads. These glass beads will provide a material that is chemically and geometrically similar across different tests. The bead particle sizes that will be used will closely resemble the particulate sizes that have been used in this paper.

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